PATENT SPECIFICATION

DRAWINGS ATTACHED

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COMPLETE SPECIFICATION

Improvements in or relating to Drive Mechanisms for Imparting Intermittent Rotary Motion to a Shaft

We, SAMUEL GRIFFITHS (WILLENHALL) LIMITED, a British Company of Bentley Works, John Harper Street, Willenhall, in the County of Stafford, do hereby declare the invention for which we pray that a Patent may be granted to us and the method by which it is to be performed to be particularly described in and by the following statement:—

This invention relates to power presses of the kind comprising a driving shaft (herein for convenience called "the crankshaft") having a crank, cam, or other eccentric element, and further comprising a motor driven rotary power storage member (usually in the form of a flywheel) for imparting the requisite motion to the crankshaft, and a ram for carrying one of a pair of co-operating tools, which ram is driven from a crankshaft and guided for movement towards and away from the other of the pair of tools. Such power presses are hereinafter referred to as being of the kind specified.

It is to be understood that the term "press" is not intended to be limited in its scope to machines of the type which press or stamp material to a desired shape, but is also intended to include within its scope other like machines, that is to say machines including a pair of relatively reciprocable tools. Examples of such other machines include press brakes, i.e. presses having a long bed and which are used for bending work, and cutting or shearing apparatus.

In our co-pending application No. 44493/64, dated 31st October 1964, (1,103,599), we have described and claimed a power press of the kind specified wherein the crankshaft is driven through the intermediary of a differential gear which comprises a rotary planet carrier having two sets of planetary gears meshing with respective sun gears of which one is connected with an output shaft and the other is connected with a shaft co-axial with the output shaft, and wherein rotatable elements of respective brakes are connected with the output shaft and co-axial shaft and co-operate with non-rotatable elements movable relatively into or out of co-operation therewith under the control of brake actuating means which serve selectively to apply either of the brakes while releasing the other so that the output shaft is either driven or braked as required.

In many cases the rate at which a power press is required to work depends on the particular operation to be performed. In the power press described and claimed in our co-pending application aforesaid the differential gear performs the dual function of a clutch and brake unit and a gearbox providing a predetermined step down velocity ratio. This ratio, however, cannot be varied without either exchanging the differential gear as a whole for another similar gear affording a different ratio, or disassembling the differential gear and exchanging the gear elements thereof for other such elements to provide a different ratio. This procedure is not suitable for general industrial application due to the difficulties associated therewith under workshop conditions.

The object of the present invention is to provide a power press of the kind specified having a variable rate of working which overcomes the above mentioned disadvantages.

According to the present invention we provide a power press of the kind specified wherein the crankshaft is driven through the intermediary of a differential gear which comprises a rotary planet carrier having at least three sets of planetary gears meshing with respective sun gears, one of the latter being connected to an output shaft and the others being connected to respective shafts co-axial with said output shaft, and wherein respective brakes are associated with the output shaft and each of the co-aixal shafts, said brakes each having a rotatable element connected to the respective output shaft or co-axial shaft and a non-rotatable element movable relatively into or out of co-operation with its respective rotatable element under the control of brake actuating means which serve selectively to apply any one of the brakes while releasing all of the others so that the output shaft is either braked or driven at one of at least two speeds as required according to which of the brakes is applied, the sets of planetary gears associated with the co-axial shafts being arranged on both sides of the set associated with the output shaft, and there being at least one such set on each side of the output shaft planetary gears and a corresponding number of co-axial shafts extending outwardly from each side of the planet carrier.

It will be understood that the output shaft may either drive the crankshaft of the press through further gears, or may itself comprise the crankshaft, or an extension thereof.

With the power press according to the invention the gear ratio between the motor and the crankshaft can be varied by simply actuating the appropriate one of the brakes operating on the coaxial shafts. In particular the gear ratio can be altered while the press is in motion so that it is possible to vary the speed of the stroke of the power press within a single cycle. This is particularly useful where a large number of operations require to be performed, particularly on stock being fed automatically through the press, and where these operations are of such a nature (e.g. deep drawing as to require comparatively slow relative movement between the pair of tools carried by the press while they actually contact stock. In such a case the press can be arranged to operate in a cycle which starts at top dead centre at high speed, changes to slow speed as the reciprocating tool contacts the stock, and reverts to high speed after the operation is performed. In this way the repetition rate of the press can be materially increased so that its throughput is likewise increased while it still satisfactorily performs slow operations.

Even if it is not required to vary the speed of operation within the cycle, such a press still has considerable advantages in that it can be readily used for a wider variety of operations than conventional power presses.

Preferably the planet carrier is so dimensioned as to have a sufficiently large moment of inertia to serve as a flywheel.

The co-axial shafts may each be formed or provided with a radial flange carrying the rotatable element of the respective co-axial shaft brake externally of the planet carrier.

Preferably the output shaft brake is applied or operated by spring means and arranged to be retracted or disengaged by the establishment of fluid pressure in a main brake device.

The co-axial shaft brakes may be so arranged as to be applied or operated upon esteablishment of fluid pressure in respective auxiliary brake cylinders, the brake actuating means being so arranged that pressure is established in said main brake cylinder wheneven pressure is established in any of the auxiliary brake cylinders.

In an alternative arrangement, the sets of planet gears associated with the co-axial shafts may all be disposed on the same side of the set of planet gears associated with the crankshaft, the respective co-axial shafts all emerging from the planet carrier at the same side.

The invention will now be described by way of example with reference to the accompanying drawings wherein:—

FIGURE 1 is a view in front elevation of one construction of two speed press in accordance with the invention,

FIGURE 2 is a fragmentary view partly in section on the line II—II of Figure 1 showing the differential gear unit and the crankshaft drive,

FIGURE 3 is a radial section through the upper rearward half of the differential gear unit shown in Figure 2,

FIGURE 4 is a radial section through the upper forward half of the differential gear unit shown in Figure 2, and

FIGURE 5 shows schematically a suitable pneumatic control circuit.

Referring to Figure 1, the body 10 of the press is formed as a frame in the rectangular aperture 17 of which is mounted a bed 18 at its lower end. The aperture 17 is provided with vertically extending guideways 20 in which a ram 21 is mounted so as to be capable of reciprocating towards and away from the bed 18.

Referring now to Figure 2, such reciprocatory motion is transmitted to the ram 21 from a crankshaft 22 which incorporates or carries an eccentric element such as a crank 23 operatively connected with the ram 21 through the intermediary of a connecting rod 24, the lower end of which is coupled to the ram 21 by a ball and socket joint.

The upper end or head 19 of the press body is of hollow form and at its forward and rearward sides carries respective bearings 26 and 27 which accommodate the forward

and rearward end portions of the crankshaft 22, the latter extending fore-and-aft through the head 19. The head also accommodates the mechanism for transmitting drive from a driving motor 28 (Figure 1) to the crankshaft 22. The driving motor 28 is conveniently mounted on top of the side members of the body 10 above the head 19. Such mechanism includes a differential gear unit 25 mounted fore-and-aft in the head 19 above the crankshaft 22, the output or main shaft 29 of which drives the crankshaft via a pinion 14 meshing with a gear wheel 15 disposed at the rear side of the body. The drive mechanism also includes a main brake unit 11 and two auxiliaray brake units 16 and 16a.

Referring now to Figures 3 and 4, differential gear unit 25 is of the epicyclic type. This gear unit includes a planet carrier 33 in the form of a generally cylindrical casing having a circumferential wall 34 incorporating externally a plurality of V-shaped grooves 31 for receiving 'V' belts 35 engaging in a corresponding number of grooves of a driving pulley 36 on the shaft of the motor 28.

The planet carrier 33 has end walls 39 and 40 which are centrally apertured and furnished with sealing members 41 and 42 at their inner boundaries.

A plurality of sets of planet gears are supported from, and disposed internally of, the planet carrier 33. Typically each set of planet gears includes three such gears at equal angular intervals about the axis of the main shaft 29. There are three sets of planet gears, only one gear of each set being seen in Figures 3 and 4. Three planet gears 43, 44 and 44a, one from each set, having different diameters and numbers of teeth respectively are keyed to a common planet shaft 45 supported at opposite ends by roller bearing assemblies 46 and 47 in the respective end walls 39 and 40 of the planet carrier, while balls 48 and 49 interposed between bearing recesses in the ends of the shaft 45 and thrust members in the end walls of the planet carrier take any axial thrust. Two other similar shafts (not shown) each also carry one gear from each set.

The planet gears 43, 44 and 44a are keyed to the shaft 45 by keying material 50 preferably of non-metallic form. Such material may be a synthetic resin having the requisite mechanical strength and capable of being injected into communicating cavities in the shaft 45 and in the planet gears 43, 44 and 44a respectively, which latter may be in the form of axially extending grooves and separated by intervening serrations or ribs, the material being thereafter caused or permitted to harden. For this purpose the shaft 45 is formed with an axial bore 51 and radially extending bores, of which two are shown at 53 and 53a, leading to the communicating cavities in which keying portions of the material are disposed.

A synthetic resin such as nylon provides some degree of elasticity with respect to the transmission of torque between each planet gear 43 and the shaft 45. This factor taken together with ability to inject the keying material whilst the planet gears of each set are brought fully into mesh with their respective sun gears, hereinafter mentioned, ensures that each planet gear of a set transmits an equal or approximately equal proportion of the torque load required to be transmitted from the planet carrier 33 to the main shaft 29. Further the dimensions of the keying portions may be made such that should any failure take place it does so by shearing of the keying portions rather than by stripping of the meshing teeth of the planet and sun gears or failure of other parts which are more expensive to replace than the keying material. This form of keying may be applied either to all the gears in each set of planetary gears or to only two out of the three in each set.

The central set of planet gears, one of which is shown at 43, meshes with a sun gear 54 which is secured to the main shaft 29 by means of a key 56.

The outer sets of planet gears, respective ones of which are shown at 44 and 44a, mesh with respective co-axially mounted sun gears 58 and 58a which are fixedly mounted on co-axial tubular shafts 59 and 59a by means of keys 60 and 60a. Such tubular shafts terminate in respective attachment flanges 61 and 61a at their ends axially outward of the planet carrier, and pass through the sealing members 41 and 42 carried by the end walls of the planet carrier 33.

The planet carrier 33 forms the input element of the epicyclic gear unit and the main shaft 29 and aubular shafts 59 and 59a carrying respectively sun gears 54, 58 and 58a form respectively an output element and first, and second control elements. The overall ratio afforded between the input element and the output element, when one of the control elements is held stationary, is determined by the difference of the ratio afforded between the planet gear 43 and its companion sun gear 54 on the one hand, and on the other hand that afforded between either the planet gear 44 and its companion sun gear 58 or the planet gear 44a and its companion sun gear 58a, according to which of the sun gears 58 and 58a is held stationary. This form of gear enables both the two overall reduction ratios and also the ratio of these reduction ratios to be selected from a very wide range without any, or any substantial, alterations in the design and overall size of the gear. Typically the overall ratios may be in the range 1:3 to 1:12, or thereabouts, to give high speed and low

speed operation of the ram, the ratio of the ratios being typically 3:1 or 4:1. However, special requirements as to very high speed or very low speed working can be met by selecting the overall ratios within a range 1:1.8 to 1:272 or thereabouts and employing either a low value for the ratio of the ratios so as to obtain two high speeds or two low speeds, or high value of the ratio of the ratios so as to obtain facilties for very high speed and very low speed working in the same press.

The planet carrier also serves as a flywheel and is in all cases driven to rotate at a sufficiently high speed to provide substantially the whole of the energy required for the working stroke of the press (typically at 500 to 600 rpm) and at the same time enable the diameter of the planet carrier to be kept to compact dimensions. This enables the epicyclic gear to be accommodated in the head 19 without the cross-section dimensions of the latter in a vertical plane extending transversely of the press exceeding the overall lateral dimensions of the press as determined by the frame.

The planet carrier 33 is located radially with respect to the tubular shaft 59 and 59a by ball bearing assemblies 63 and 63a.

The auxiliary brake units 16 and 16a are arranged respectively to hold stationary the tubular shafts 59 and 59a. These units are described below.

To the flanges 61 and 61a of the tubular shafts 59 and 59a are secured respective hubs 64 and 64a which are splined externally as seen at 65 and 65a and carry rotatable brake elements in the form of generally annular plates 66 and 66a splined internally to cooperate with their respective splines 65 and 65a so as to be slidable therealong and equipped with friction linings 67 and 67a.

Associated with these rotatable brake elements are fixed brake elements comprising annular chambers 68 and 68a which each at their axially outward ends have a respective attachment flange 69 and 69a secured to forward and rearward closure members 70 and 70a received in apertures 30 and 30a in the front and back walls 12 and 13 of the head 19. The closure members are centrally apertured and carry roller bearing assemblies 32 and 32a in which the main shaft 29 is supported. At their axially inner ends the chambers 68 and 68a each have a radially inwardly projecting flange 71 and 71a. Internally the chambers 68 and 68a are splined, as indicated at 72 and 72a and the outer peripheries of rotationally fixed annular plates 73 and 73a are shaped to engage slidably with these splines, the plates 73 and 73a being interleaved with the plates 66 and 66a respectively and their linings.

These two brake assemblies are operated by piston and cylinder assemblies 79 and 79a respectively. The piston and cylinder assembly 79 comprises an annular outer component 80 which is of stepped form in transverse cross-section. An inner stepped member of annular form serves as an annular piston 81, which in combination with the outer component 80 defines an annular chamber 82, suitable seals 83 and 84 being secured respectively to the outer component, or cylinder 80 and to the piston 81 by screws 85 and 86.

The piston 81 serves to compress the assembly of rotatable and non-rotatable brake plates 67 and 73 through the intermediary of a plurality of cylindrical thrust elements, one of which is shown at 90. These thrust elements are slidably mounted in openings in the forward closure member 70. At its inner periphery the piston 81 includes a radially projecting flange 87 which is pressed axially forwardly by means of a plurality of angularly spaced coiled compression springs 88 accommodated in sockets 89 formed in the closure member 70 and bearing against the flange 87. When air under pressure is admitted to the chamber 82 the piston 81 is moved axially rearwardly against the force of the springs 88 so as to compress the brake plates 67 and 73 and thereby hold the tubular shaft 59 against rotation. When the air pressure in the chamber 82 is released the tubular shaft 59 is free to rotate.

The assembly 79a is of similar construction and the above description applies equally to that assembly, similar parts shown in Figure 3 of the accompanying drawings being given the same reference numerals with the suffix a. The assembly 79a operates in relation to the tubular shaft 59a so that the latter is held against rotation when air under pressure is admitted to the chamber 82a.

The main brake unit 11 is arranged to hold the main shaft 29 against rotation when the tubular shaft brakes 16 and 16a are both inoperative, and this unit is described below.

The piston and cylinder assembly 79 also carries at its forward end a further cylindrical component 190. The latter also includes at its rearward end an internally projecting annular flange 91. The cylindrical component 190 is internally splined as indicated at 92 and engaged with such splines is a pair of annular, non-rotatable, brake plates 93 carrying friction linings 94.

The main shaft 29 carries at its forward end a further movable brake element. For this purpose the main shaft has a brake hub 74 secured thereto by splines, the hub being formed externally as a gear ring 75 which engages with the appropriately toothed inner periphery of rotatable brake plate 76. Although not shown, the rotatable brake plate 76 is formed with passageways to permit air to flow through the plate so as to cool the

latter. The rotatable brake plate 76 co-operates with the non-rotatable brake plates 93.

The cylindrical component 190 carries at its forward end an annular plate 95 affording a fowardly projecting annular shoulder 96. An axially extending outer flange 98 of an end member 97 is located slidably on the shoulder 96 and an expansible air tube 99 of annular form is arranged between the plate 95 and the end member 97. The end member 97 carries fixed thereto an axially rearwardly extending generally conical sleeve part 100 having at its rearward end a radially outwardly extending flange 101. The rear end of the sleeve part 100 extends around the forward end of the brake hub 74 with a clearance and the flange 101 is in register with the brake plates 93. A plurality of pairs of angularly spaced coiled compression springs 102 are arranged between the plate 95 and the flange 102 so that the latter normally presses the brake plates 93 and 76 into contact and the main shaft 29 is thereby held against rotation.

Admission of air or other fluid under pressure to the tube 99 causes the main shaft brake to be released against the action of the springs 102. Simultaneous admission of such pressure fluid to either the chamber 82 or the chamber 82a then causes the sun gear 58 or 58a to be held stationary so that the main shaft 29 is rotated at a high speed or low speed accordingly.

Preferably a control system would be employed such that it was not possible for more than one of the tubular shafts 59 and 59a to be braked at the same time. Such control system may incorporate appropriate valves in the pneumatic (or hydraulic) circuits which valves are actuated either manually, mechanical interlocks being provided to prevent more than one brake being applied at any given instant, or electrically by means of microswitches or other means triggered by parts of the press itself. The latter system is especially suitable where, as previously mentioned it is desired to vary the rate of working within a single cycle of operations.

A suitable pneumatic control circuit is illustrated in Figure 6. Compressed air from a source 103 is supplied to an ON/OFF valve 104 from which a pipe 106 leads to a two way valve 105 and a branch pipe 107 leads to the main brake unit 11. The two outlets from the valve 105 are connected by pipes 108 and 108a to the auxiliary brake units 16 and 16a respectively.

Thus when the valve 104 is open the main brake is relased and one or other of the auxiliary brakes is applied so that the press is driven at the speed selected by the setting of the valve 105. When the valve 104 is closed the main brake is applied and the auxiliary brakes are both released so that the press is inoperative.

A rotary cam switch 109 driven from the crankshaft 22 is employed to change the setting of the speed control valve 105 during the working cycle of the press.

In alternative arrangements it would be possible for more than two speeds to be provided, one set of planetary gears, in addition to those meshing with the sun gear carried by the main shaft, being required for each such speed. Preferably such sets of planetary gears would be arranged on both sides of the set driving the crankshaft and a plurality of co-axial shafts would emerge from each side of the planet carrier. Each such co-axial shaft would carry a set of brake plates and a separate piston and cylinder assembly would be provided to operate each brake. It will also be possible for both or all the set of planetary gears meshing with the sun gears carries by the co-axial shafts to be disposed on the same side of the set meshing with the sun gear carried by the crankshaft.

In some circumstances it may be desirable for the one of the co-axial shafts to serve as the output shaft from which the crankshaft is driven, and in this case the main shaft and the remaining co-axial shaft or shafts would serve as the control elements and the pneumatic brake circuit would be modified accordingly.

WHAT WE CLAIM IS:—

1. A power press of the kind specified wherein the crankshaft is driven through the intermediary of a differential gear which comprises a rotary planet carrier having at least three sets of planetary gears meshing with respective sun gears, one of the latter being connected to an output shaft and the others being connected to respective shafts co-axial with said output shaft, and wherein respective brakes are associated with the output shaft and each of the co-axial shafts, said brakes each having a rotatable element connected to the respective output shaft or co-axial shaft and a non-rotatable element movable relatively into or out of co-operation with its respective rotatable element under the control of brake actuating means which serve selectively to apply any one of the brakes while releasing all of the others so that the output shaft is either braked or driven at one of at least two speeds as required according to which of the brakes is applied, the sets of planetary gears associated with the co-axial shafts being arranged on both sides of the set associated with the output shaft, and there being at least one such set on each side of the output shaft planetary gears and a corresponding number of co-axial shafts extending outwardly from each side of the planet carrier.

2. A power press according to claim 1 wherein the planet carrier is so dimensioned as to have a sufficiently large moment of inertia to serve as a flywheel.

3. A power press according to claim 1 or claim 2 wherein the co-axial shafts are each formed or provided with a radial flange carrying the rotatable element of the respective co-axial shaft brake externally of the planet carrier.

4. A power press according to any one of the preceding claims wherein the output shaft brake is applied or operated by spring means and arranged to be retracted or disengaged by the establishment of fluid pressure in a main brake device.

5. A power press according to Claim 4 wherein the co-axial shaft brakes are so arranged as to be applied or operated upon establishment of fluid pressure in respective auxiliary brake cylinders, the brake actuating means being so arranged that pressure is established in said main brake cylinder whenever pressure is established in any of the auxiliary brake cylinders.

6. A power press of the kind specified having a driving mechanism substantially as hereinbefore described with reference to and as shown in Figures 3 and 4 of the accompanying drawings.

7. A power press substantially as hereinbefore described with reference to and as shown in the accompanying drawings.

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1192544 COMPLETE SPECIFICATION

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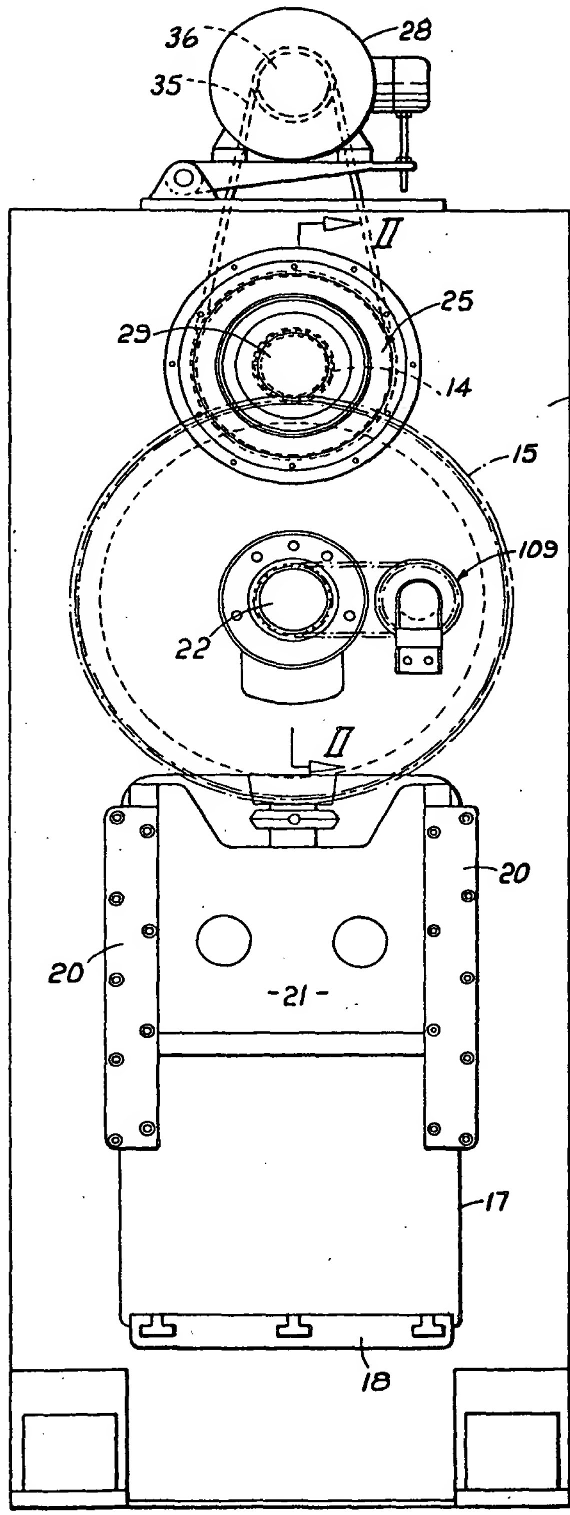
Sheet 1

Fig.1.



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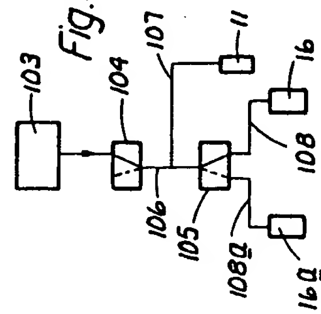
Sheet 2

Fig. 5.



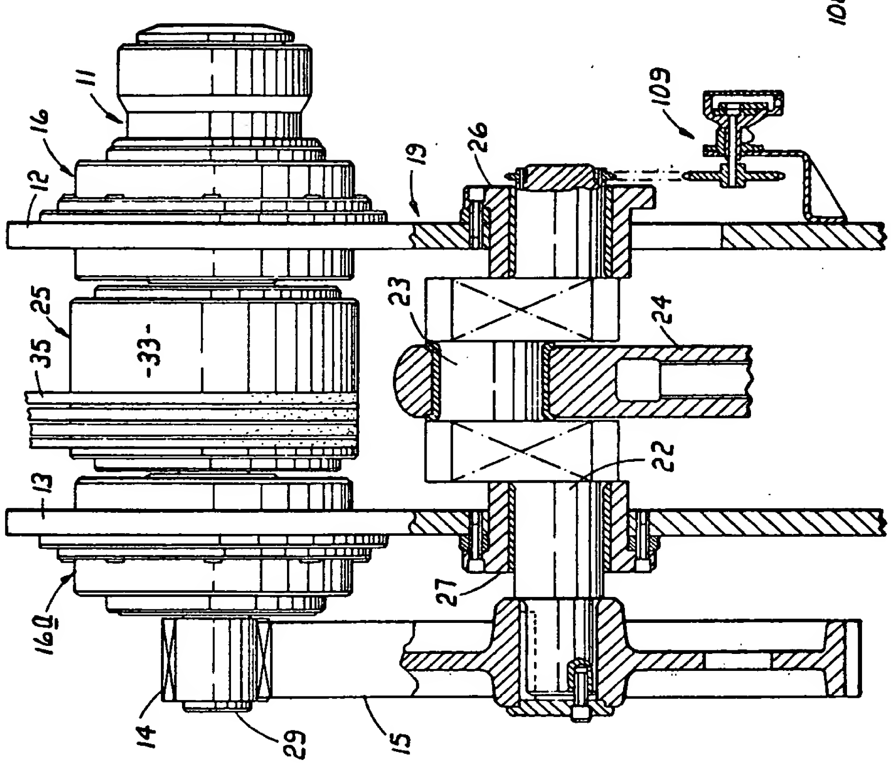


Fig. 2.

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Sheet 3

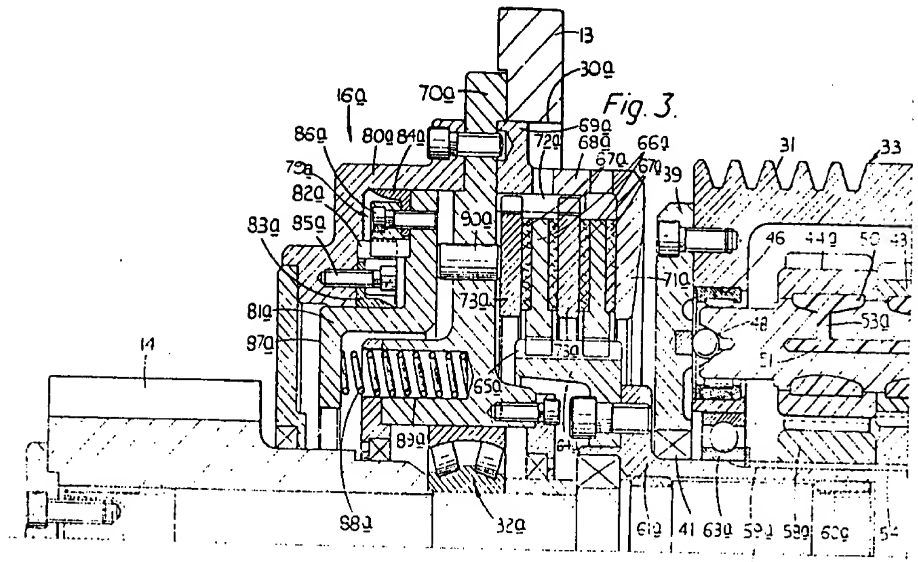

Fig. 3.

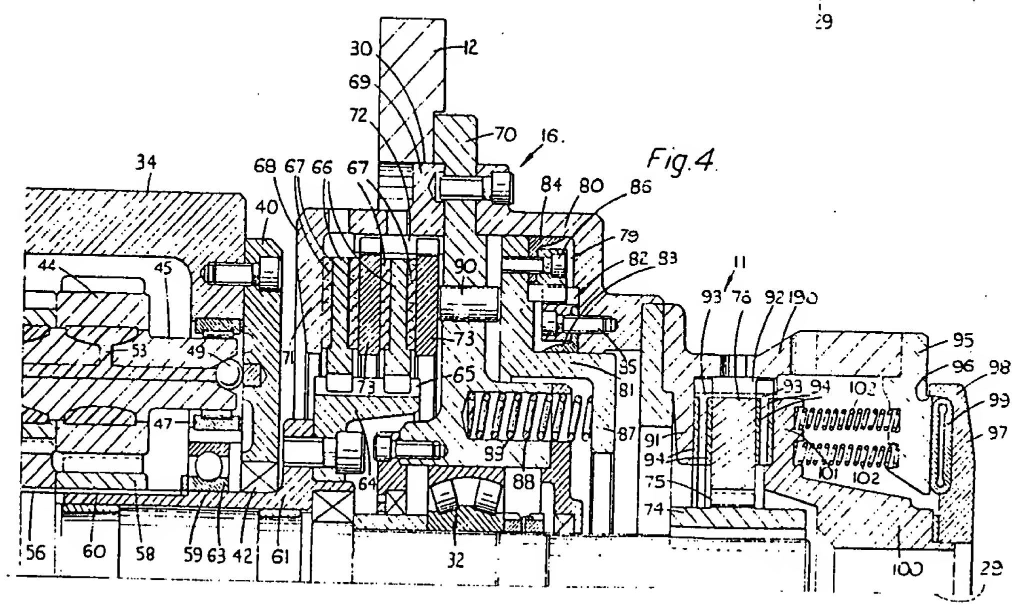

Fig. 4.